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Mechanical Performance of a Two Stage Centrifugal Compressor under Wet Gas Conditions

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ABSTRACT

As subsea compression becomes a vital technology to the successful production of gas reserves in the North Sea, several technology issues will come to the forefront of the oil and gas industry. One of these important subjects is the capability to compress gas which includes a significant amount of liquids. Compressing wet gas requires knowledge in areas such as the prediction of turbomachinery performance with the mixed phase flow as well as the mechanical reliability of machinery in the same environment. This paper presents experimental results from a wet gas test campaign which, among other goals, is focused on characterizing the mechanical performance of a two stage compressor operating under wet gas conditions. Various mechanical parameters are monitored in the test program including rotor radial and axial vibration, rotor thrust, and shaft torque. A full array of wet gas conditions are tested with a suction pressure of 20 bar (300 psia) and liquid volume fractions in the range of 0.5 to 5%. The operating fluids are air and water, and the two stage compressor is operated at three speed lines ranging from high flow to low flow conditions. Significant variations are noted in the axial thrust, axial vibration and shaft torque. Thrust variations range from seemingly neutral thrust conditions at very low water injection rates to significant thrust increases (as compared to dry condition) for very high water injection rates. Rotor axial vibration is characterized by large amplitude and very low frequency, especially for the case in which the rotor thrust is balanced by the water injection. During higher levels of water injection, rotor axial vibration is generally characterized by relatively large amplitude and slightly higher frequency, although still very low as a percent of running speed. Variations in radial vibration are also noted, but to a much lesser extent.

INTRODUCTION

As subsea compression becomes a vital technology to the successful production of gas reserves in the North Sea, several technology issues will come to the forefront of the oil and gas industry. One of these important subjects is the capability to compress gas which includes a significant amount of liquids. Compressing wet gas requires knowledge in many areas such as the prediction of turbomachinery performance with the mixed phase flow as well as the mechanical reliability of machinery in the same environment. In many turbomachinery designs, it is common to use process fluids for ancillary purposes such as shunt injection, cooling flow, etc. In most land based applications, this is a relatively simple endeavor as the space required for the conditioning of this extraction flow is readily available, and large items such as liquid/ gas phase separators can be easily incorporated into the design. As machinery is moved off-shore, this real-estate becomes more expensive, and is exceedingly expensive for subsea applications. Therefore, it is advantageous to incorporate designs which allow for the extraction of clean, dry gas that won't require a significant increase in the footprint of the machinery skid or package.

In response to this market demand, technology for wet gas compression is being developed for subsea applications. This development requires a comprehensive look at all operational parameters or a centrifugal compressor in wet compression. This paper focuses on the mechanical aspects of compressor operation including rotor radial and axial vibration, rotor thrust, and shaft torque. A description of the test program, test rig, thrust and vibration measurements, and discussion of the resulting trends is presented. Lastly, conclusions about the mechanical performance and considerations for future wet compression designs are discussed. on dynamic characteristics of compressors and turbines.

BACKGROUND

There have been primarily two other research efforts which have investigated and reported on the effects of wet gas on centrifugal compressor mechanical and thermodynamic performance. The first is reported by Brenne et. Al. The authors completed a series of tests on a centrifugal compressor with gas volume fractions ranging from 1.0 to 0.97. The test compressor is a single stage compressor driven by a 2.8 MW motor at speeds from 6,000 to 13,000 rpm with suction pressure ranging from 30 to 70 bar. The test is completed with a mixture of gas and liquid hydrocarbons.

Aerodynamic/thermodynamic and rotordyanmic results are reported for this research. Since the focus of this paper is the compressor mechanical performance, the authors will focus on this aspect in the reported work. The compressor vibration is measured with proximity probes at both the drive and non drive ends of the compressor in the horizontal and vertical direction. In addition, axial vibration measurements are made. The vibration spectrum of the compressor during both wet and dry operation is found to be virtually the same in the majority of the tests. At one test condition, subsynchronous vibrations are experienced during wet compression which suggested rotor instability. However, the primary vibration peaks remained unchanged from the dry tests. The authors conclude that vibration levels of the compressor are not significantly affected by the presence of liquid when it is injected in a uniform pattern. They also comment that the vibration patterns may change between dry and wet operation if a non-uniform injection profile is used or for low gas quality [1]

Fabbrizzi et. Al developed a single stage open loop test stand which is used to investigate the effects of wet compression based on variations in the liquid mass fraction (LMF), impeller speed, droplet size, and the injection pattern of the liquid. An air-water mixture is flowed through the compressor with an ambient suction pressure and LMF from 0 to 0.6. The authors reported results of the testing related to aerodynamic/thermodynamic performance of the compressor, but no results related to mechanical performance were provided in the current publication [2].

In both test campaigns described above, a focus is placed primarily on the aerodynamic and thermodynamic performance of the compressors. While, this is a critical aspect of understanding wet compression, it is also important to further investigate the mechanical effects of wet compression.

DESCRIPTION OF TEST PROGRAM AND TEST SET-UP

In order to characterize the wet compression performance of a two stage centrifugal compressor a test loop is developed and constructed. This is accomplished with an integrated air and water test loop. Instrumentation is placed on the loop in order to meet ASME PTC 10 requirements for dry performance measurements and quantify the properties of the liquid being injected into the air flow.

In the design of the test set-up, a focus is placed on the mechanical operation of the compressor with wet gas. This included modification of the thrust bearing housing to accommodate thrust measurements, inclusion of proximity probes for four radial and one axial vibration measurement, and a shaft torque measurement.

During testing the test compressor is challenged with various mixtures of air and water on three different speed lines. Measurements are taken at speeds of 8,000, 9,500, and 11,000 RPM. A mixture of high pressure air and water is flowed through the test loop with liquid volume fractions varying from 0.5% to 5%. The suction pressure of the compressor was held within 19 and 20 bara. A detailed description of the test set-up is provided below.

LOOP OVERVIEW

The Process and Instrumentation Diagram (P&ID) of the test loop is shown in Figure 1. There are two main interconnected loops shown in this diagram: the air loop (highlighted in green) and water loop (highlighted in blue). An overview 3D rendering of the test set-up is show in Figure 2, a picture of the outside test loop components is shown in Figure 3, and a picture of the inside test loop components is shown in Figure 4. In the 3D rendering, the air loop has white piping and the water loop has gray piping. The water loop and air loop systems are described below in more detail.

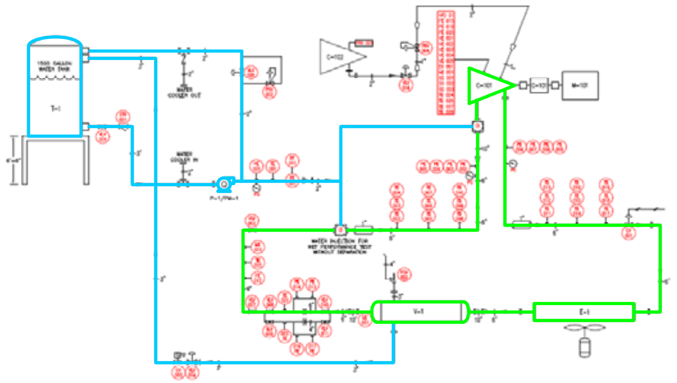

Figure 1 - P&ID of wet gas compression test loop

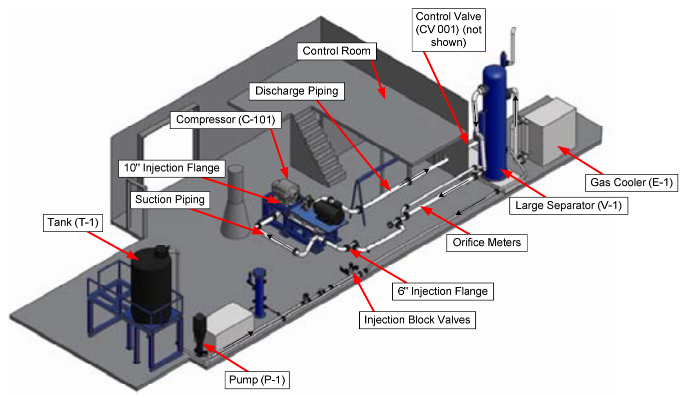

Figure 2 - Graphical Overview of Test Set-up

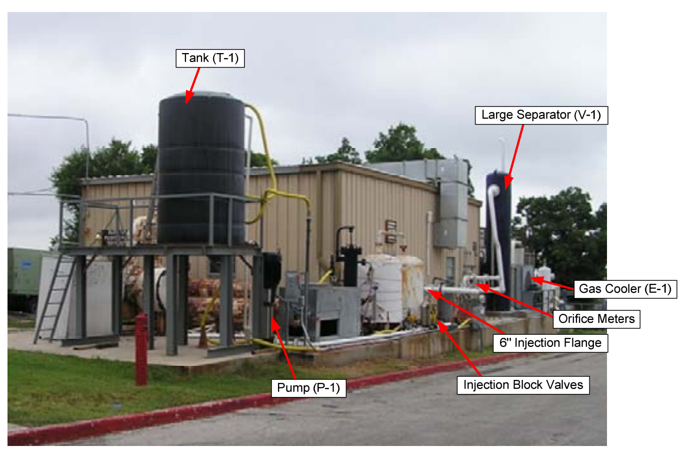

Figure 3 - Overview of Test Loop Components Outside of Test Building

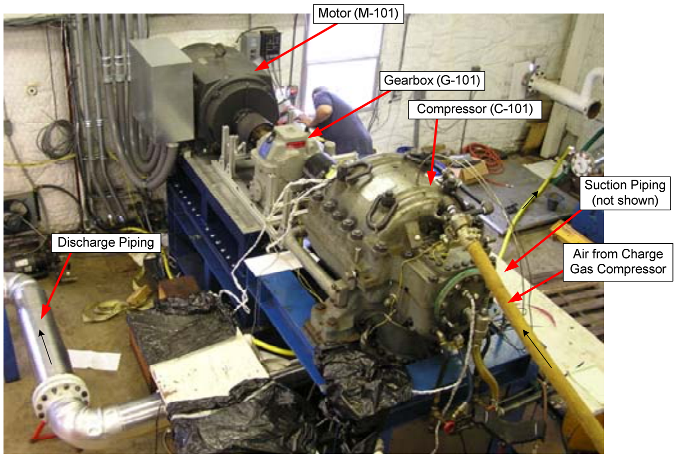

Figure 4 - Overview of Test Loop Components Inside of Test Building

AIR LOOP

The air loop is comprised of a closed test loop which has a centrifugal compressor, gas cooler, large separator, and various instruments.

Centrifugal Compressor

On the upper right-hand corner of the P&ID and in the middle of the 3D rendering (Figure 2) is the centrifugal compressor (C-101). The test compressor is a 1M4-2 Clark centrifugal compressor. This compressor is connected to a motor through a gearbox in order to achieve a maximum operational speed of 14,000 RPM. A picture of the compressor drive train is shown in Figure 5.

The compressor has two stages of compression. Its maximum continuous speed is 14,280 RPM with the first critical speed occurring around 8,500 RPM. The casing of the compressor is designed to have a maximum inlet pressure of 20.7 barg (300 psig) and maximum discharge pressure of 34.5 barg (500 psig). The compressor rotor is supported by two tilt pad (on-pad) bearings. The thrust bearing (installed on the non-drive end) is also a tilt pad bearing. Several measurements are included on the centrifugal compressor: speed, lube oil temperature and pressure, radial and axial vibration, shaft torque, thrust, and bearing temperature.

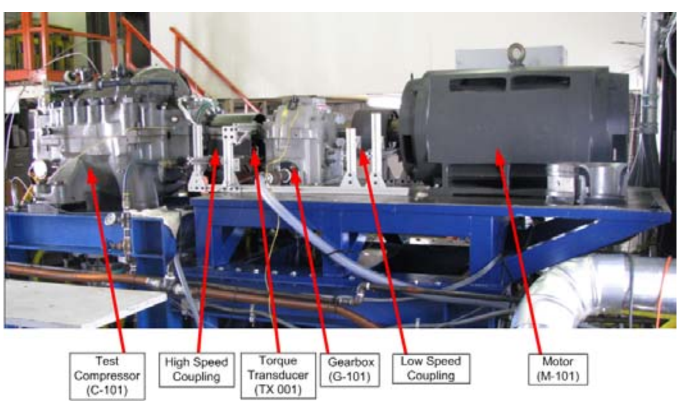

Figure 5 - Test Compressor Drive Train

Radial and Axial Vibration Measurement

The rotor vibration is measured at both ends of the compressor case. The measurement of rotor vibration is made possible through the use of proximity probes. A total of five Bentley Nevada probes are used with an output a voltage of ± 20 V. All proximity probes are gapped to approximately 50 mils and are calibrated using a GE Energy TK3-2E Proximity Probe Calibration Instrument to be linear between ± 15 mils.

A vertical probe and a horizontal probe are used on the drive end of the rotor as shown in Figure 6. Vertical, horizontal, and axial probes are installed on the non-drive end of the rotor. These proximity probes are mounted to the thrust bearing housing as shown in Figure 7.

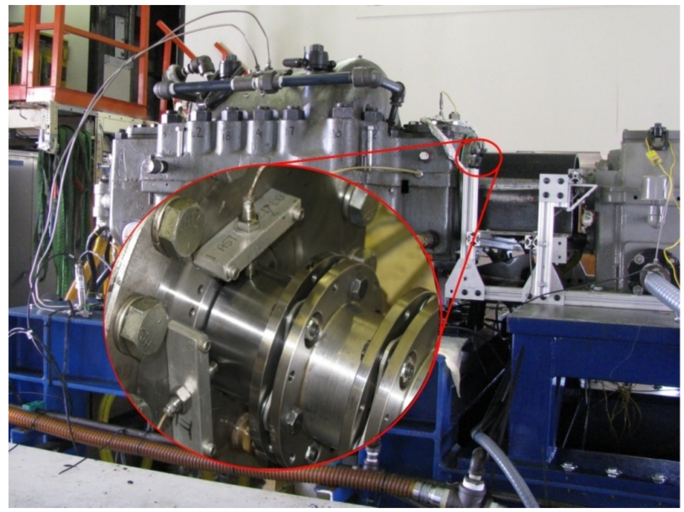

Figure 6 - Drive End Vertical and Horizontal Proximity Probes

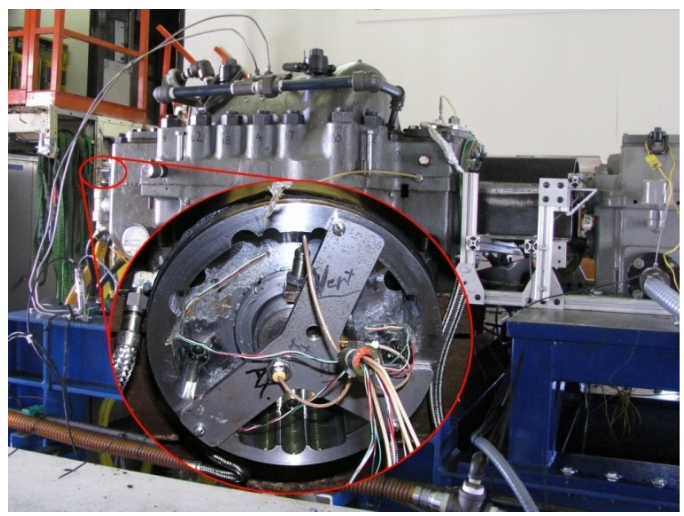

Figure 7 - Non-Drive End Vertical, Horizontal, and Axial Proximity Probes

Shaft Torque

The torque is measured on the shaft from the gearbox to the test compressor. The power required for compression is calculated with the measured torque and compressor speed. A HBM T-40 Torque Flange is used to measure the torque. The torque meter is rated to 500 N-m and has a 0 to 10 volt output with a stated sensitivity of ±0.1 volts. Figure 8 shows the torque meter installed on the gearbox shaft.

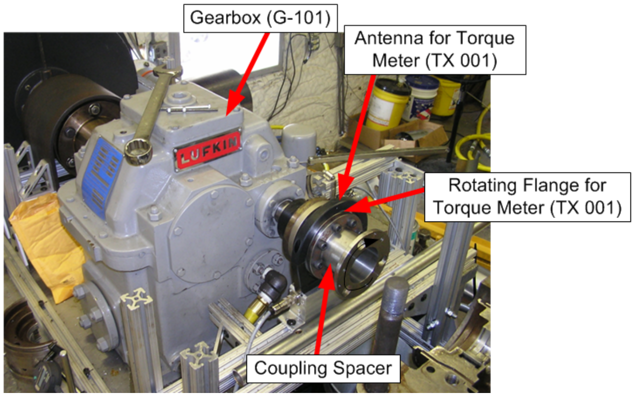

Figure 8 - Torque Meter

Thrust

The thrust of the compressor rotor is measured by redesigning the original thrust bearing housing to incorporate 3 equally distributed load cells. The housing is modified such that the NDE bearing would mate to a load washer that would then transfer the load to the load cells and on to the housing. Great care is taken to ensure the bearing clearance remained within the manufacturer specifications as well as to ensure the washer face is parallel to the thrust disc. The completed instrumented thrust bearing housing without the load washer can be seen below in Figure 9. This is accomplished by allowing tolerance for precision disc shims which are used to achieve proper alignment. A finite element analysis (FEA) is performed on each piece to verify a safety factor of two compared to the thrust bearing pads. This ensures any failure due to an excessive amount of thrust would occur at the bearing pads. The load cells used for the thrust measurement are Sensotech Subminiature Load Cells Model LFH-7I with a range of 3000lbs each.

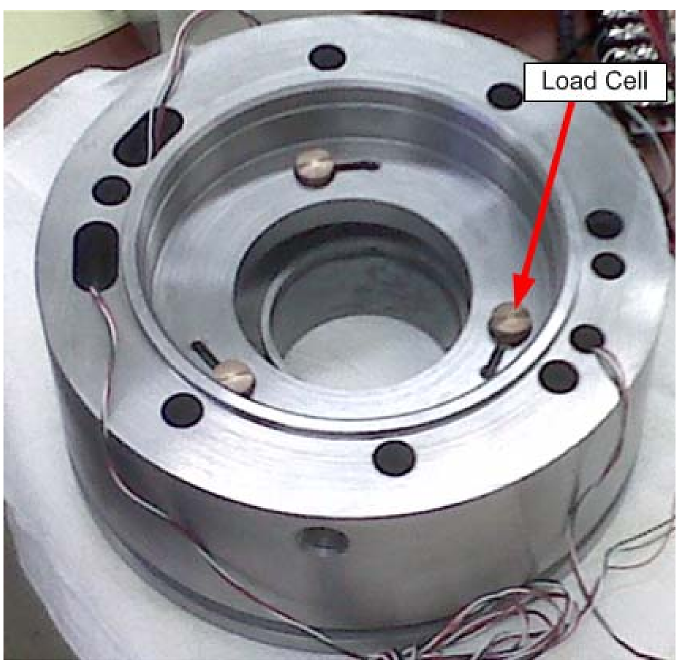

Figure 9 - Redesigned Thrust Bearing Housing Showing Distributed Load Cells

Injection Flange

At the suction flange to the centrifugal compressor (C-101) is the 10 inch injection flange (Cl) (see Figure 10). This is the location where the water is injected into the suction piping of the air loop during testing. The injection flange is comprised of eight equally spaced nozzles. The flow rate of through the injection flange is controlled by the number of nozzles in use and the discharge pressure of the pump in the liquid loop (reviewed later). The nozzles are sized in order to ensure that atomization is occurring at the exit of the nozzle.

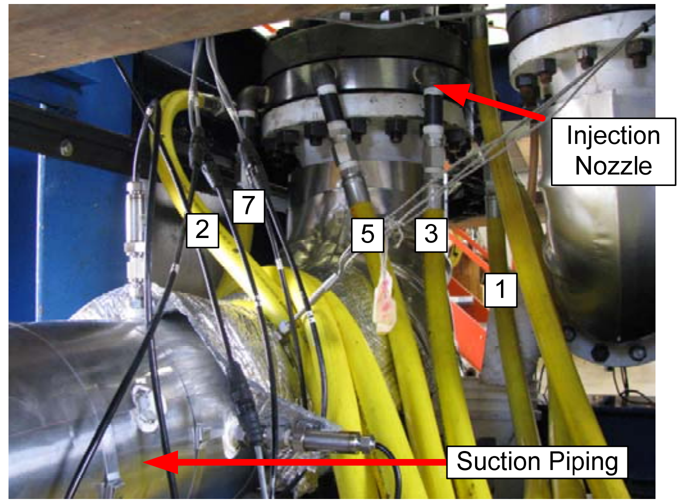

Figure 10 - Water injection flange on suction of compressor

Pressure and Temperature Measurements

Several measurements are made on the air test loop to determine the performance of the compressor. These measurements were made in accordance with ASME PTC 10 for dry performance tests. Immediately upstream of the injection flange on the 6” pipeline are the pressure and temperature measurements for the suction conditions. Temperature measurements are made at three locations on the upstream piping. At each location a temperature profile is obtained with three thermocouples.

The pressure is measured downstream of the temperature immediately before the flow enters the compressor. This measurement consists of an array of four pressure transducers all in the same radial plane spaced 90 degrees apart. The section of piping which houses these temperature and pressure measurements up to the inlet of the compressor has 1” insulation. This allows the flow to reach thermal equilibrium while minimizing the effects of heat transfer to and from the piping during testing.

Immediately downstream of the centrifugal compressor on the 6” discharge piping are pressure and temperature measurements to measure discharge conditions. Again, there are temperature measurements at three downstream locations with three thermocouples each and an array of four pressure measurements in the same radial plane immediately downstream of the compressor discharge.

Air Suction Pressure, Flow Rate, and Temperature Control

Downstream of the discharge measurements on the 6” line is a control valve. This valve is used to control the flow rate of air through the test loop and also reduce to pressure below 20.7 bara (300 psia) for the suction piping.

During the wet gas testing, air was injected into the compressor labyrinth seal region. This helps to minimize the air and water leaving the compressor through the seals. A charge gas compressor is used to supply the air and pressurize the test loop to the appropriate suction pressure before testing and also provides make-up air for the air which has leaked out of the compressor seals and diaphragm separator.

Next in the air loop is the gas cooler. This cooler is included to remove the heat in the air which was generated during gas compression. The cooling of the gas will also cause water to condense out of the air before separation.

Water Removal

Immediately downstream of the gas cooler is the large separator. The large separator is included to remove any water in the air flow during wet gas testing. The separation is important in controlling the amount of water in the air stream at the suction of the compressor.

Flow Measurement

Past the large separator is the air loop orifice bank. This is used to measure the flow rate of the air through the air loop. Two orifices are included in this bank: a 4 inch orifice and a 6 inch orifice. The instrumentation on this section included a differential pressure measurement, static pressure measurement, and temperature measurement for the 4 and 6inch orifices. During testing with a suction pressure of 20 bara, for flows from 0 to 850 am3/h, the 4” orifice plate are used. For flows from 700 to 3400 am3/h, the 6 inch orifice plate are used.

WATER LOOP

The water loop is comprised of the equipment needed to control the water flow rate into the injection flange on the air loop. The main components in this loop are the tank, pump, flow meter, and several valves.

The tank holds approximately 1500 gallons of water. The tank discharge line feeds the suction of the pump. The pump provides the required pressure boost at approximately 200 gpm to the water loop. At the discharge of the pump is a pipe tee. One leg of the tee routes water to the injection flange. On this leg there is a pressure, temperature, and flow measurement. There is also a check valve to ensure no air flows from the suction of the air loop into the water loop. Lastly, at the end of this leg, there are eight block valves which are used to control the level of flow through the injection flange.

The other leg of the tee is connected to the bypass line which re-circulates water back to the tank. The bypass line has a block valve and pressure regulating valve. The pressure regulating valve ensures the pump discharge pressure is maintained below 24.1 barg (350 psig). The bypass valve is used to control the flow rate of the water into the injection flange. Closing this valve will lead to a higher discharge pressure on the pump which will in turn cause an increase in the flow into the injection flange.

Test Results

Figure 11 represents the typical wet gas compression results recorded during the test program. Generally speaking, compared to the same dry gas condition (shaft speed and flow control valve setting), the gas flow decreases with increasing LVF and the head increases. Three speedlines are presented in Figure 11, 8000, 9500 and 11000 RPM. For the 9500 and 11000 RPM cases, the LVF limit is reached once the motor power is exceeded, resulting in a drop in shaft speed. Therefore, the open data points represent conditions recorded when the shaft speed is decreased.

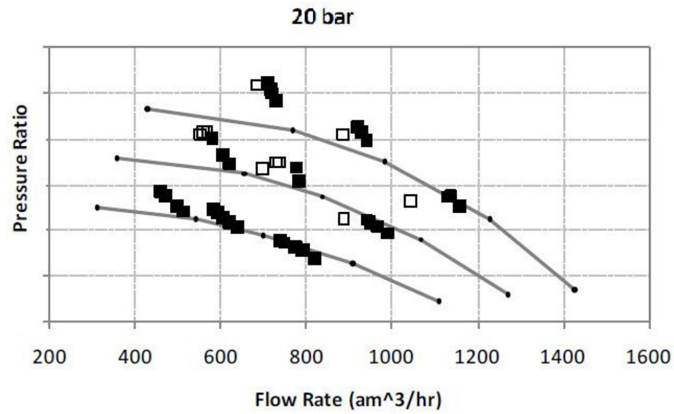

Figure 11 - Typical Wet Gas Test Results

As described above, the compressor mechanical performance data recorded during the performance testing includes mechanical vibration data (radial and axial) as well as rotor net thrust recorded at the active thrust bearing location. Figure 12 represents a typical Bode plot for the radial displacement measurements, and indicates a first critical speed at around 8,500 RPM. The second critical speed occurs at 12,000 RPM and the testing in this program is capped to 11,000 RPM. During normal, dry-compression operation, the vibration response is largely synchronous, with amplitudes less than 2.0 mils pk-pk (50 µm pk-pk) throughout the speed range.

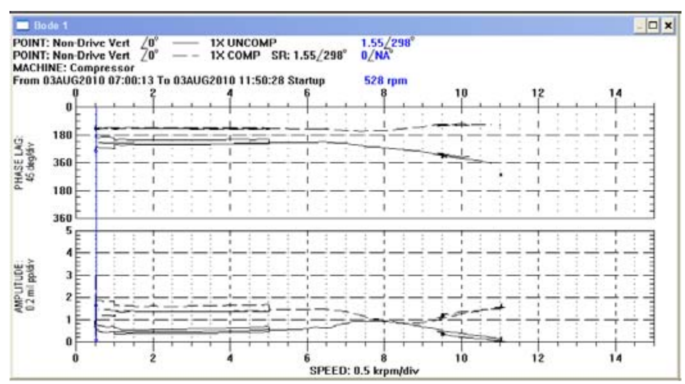

Figure 12 - Typical Bode Plot

Figure 13 is a waterfall plot for one of the wet compression tests performed at 8,000 RPM (133 Hz). Directly following the start-up, the vibration spectra shows only the synchronous response, with 2 and 3X harmonics related to the roughness at the sensor location. There is no significant sub-synchronous vibration during the dry compression periods. As the wet compression begins, there is evidence of very low frequency vibration (less than 15 Hz), which is very small in amplitude and is determined to be carry-over from the significant axial vibration described in the following paragraphs. Once the water flow is turned off, the radial vibration returns to the normal state. Throughout the entire test program, the radial vibration remained insignificant, dominated by the synchronous response.

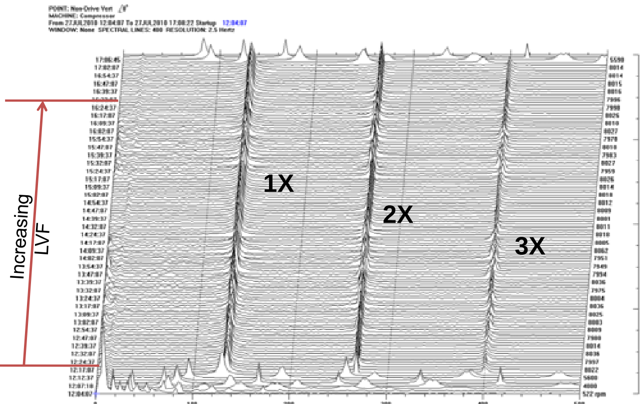

Figure 13 - Typical Radial Vibration Waterfall

The recorded axial vibration data is presented in the following figures. In the axial direction, there is a very clear change in frequency content during wet compression as compared to dry. In Figure 15 it is clear that axial vibration frequency content changes with LVF. At the beginning of the test, the axial spectra shows only 1X and the 2 and 3X harmonics, while the subsynchronous range is empty. Once the water is initiated, the low frequency axial vibration increases significantly. In fact, at the lower water flow rates, the axial motion is found to be ± 10 mils (±254 µm), the axial clearance of the thrust bearing assembly. This motion occurs at a frequency of about ½ Hz, and is later found to represent a balanced-thrust condition, and is discussed in more detail in the following paragraphs.

As the LVF increases, the very low frequency motion described above stops, and is replaced by a much lower amplitude axial vibration in the frequency range of 10-15 Hz. Initially, it is suspected that this vibration comes from a slug flow developing in the suction pipe since the injection location is 25 diameters and five turns before the compressor suction flange. In order to eliminate this potential for water collection in the suction line, the injection flange location is moved to the compressor suction flange. The results of the test are identical, thus disproving the theory regarding the slug flow generating the 15 Hz axial motion.

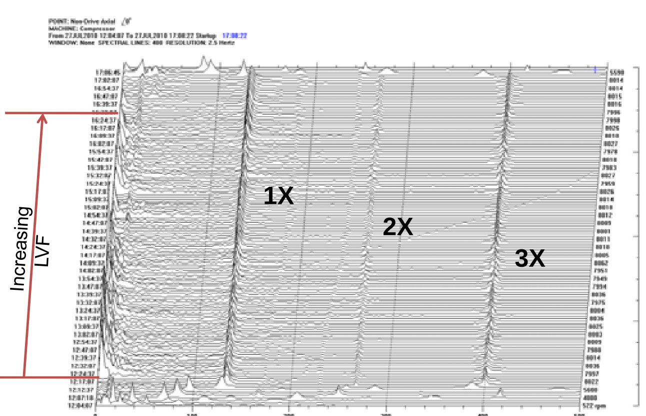

Figure 14 - Typical Axial Vibration During Wet Gas Compression

As mentioned previously, the compressor thrust bearing is modified to allow for direct measurement of rotor thrust. Figure 15 provides a sample of the measured thrust data for the 8,000 RPM speedline at three different control valve settings. Consider, for example, the CV26.5 case, which is the highest head condition used during wet gas compression testing. At LVF=0.0% (i.e. dry), the measured thrust due to dry compression is about 4 kN. Once the water is turned on to only 0.5%, the thrust actually decreases. Then, as the LVF continues to increase, so does the thrust, with a maximum value of 10 kN at 3.0% LVF. The inflection in the thrust vs. LVF curve is an unexpected result. While the decrease in thrust could be related to the impulse of the water, there should have been an off-setting increase in thrust due to the increased pressure ratio. Therefore, the measured thrust behavior warrants more investigation.

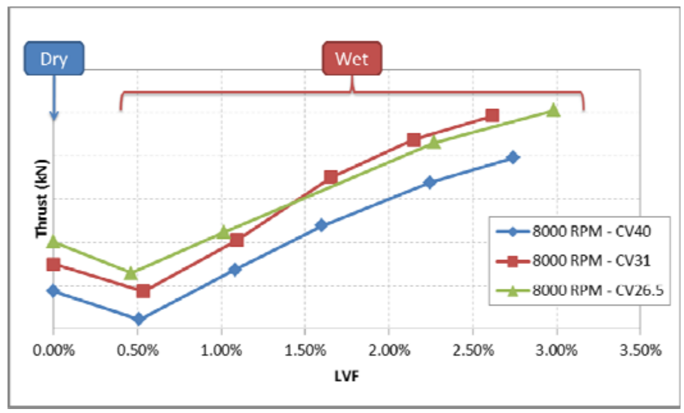

Figure 15 - Thrust vs. LVF, 8,000 RPM

In order to investigate this thrust measurement result, a simple model is developed for calculating the thrust of the rotor based on the known (or estimated) pressure conditions within the machine. Figure 16 is a schematic representing the analysis points included in the calculation. Starting with the suction pressure (P1) and the intermediate pressure between stages 1 and 2 (P2), the flow then proceeds to the stage 2 inlet (P3) and then out of stage 2 to the machine discharge (P4). Although the balance piston is connected back to suction, there is some flow resistance along the path such that the balance piston cavity pressure (P6) would be larger than P1. Finally, this machine having previously been used in oxygen service, has a rub ring on the second stage impeller which can conceivably act as an axial seal in adverse conditions, creating another cavity whose pressure is numbered P5.

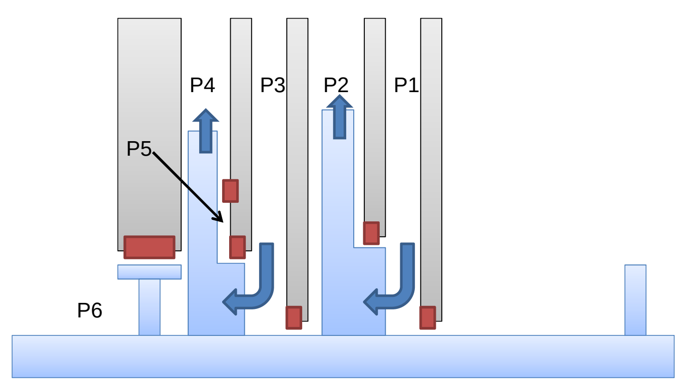

Figure 16 - Pressure-Area Forces

During normal dry compression, the model used to predict the thrust includes the flow resistance of the balance piston seal and the balance piston cavity transfer line, thus predicting P6 > P1. For the dry conditions at 8,000 RPM and CV=26.5, the calculated thrust based on measured pressures is about 4 kN (matching the measured value). This is shown in Figure 17 at the 0.0% LVF point.

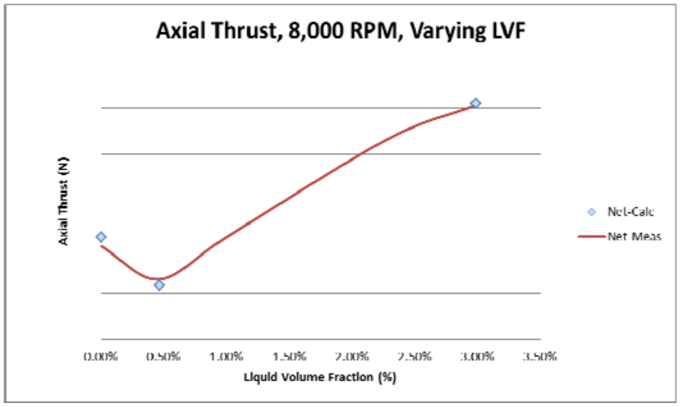

Figure 17 - Thrust vs. LVF - Calculation and Measurement

As water begins to flow through the machine, it appears as though the balance piston seal becomes a much tighter seal, such that the balance piston cavity flow rate is low enough to prevent pressure build up. In this case, the balance piston cavity equalizes with the suction pressure (i.e. P6 = P1). In this case, the net thrust is reduced, and the new calculated thrust is shown in Figure 17 at LVF=0.5%. Finally, as LVF increases, it is possible that the significant amount of water now circulating in the secondary passages alters the flow resistance of other elements as well. Post-test photographs of the rub ring region on the stage 2 impeller cover (Figure 18) suggests that there is a stagnation region in the cavity between the rub ring and the eye seal, which may occur as the rub ring becomes a higher resistance region than the eye seal. For the case of the test compressor, the eye seals are old and worn out, having clearances 2X typical. Assuming that the cavity pressure then reduces, we state that P5 = P3, and the resulting thrust calculation for LVF=3.0% is shown in Figure 17.

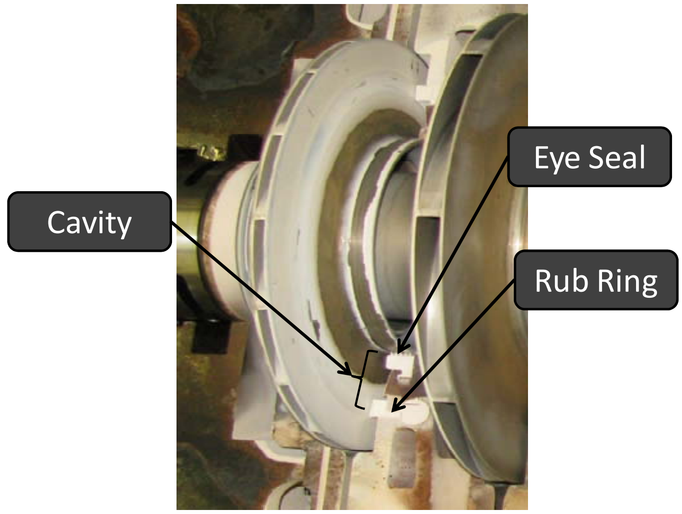

Figure 18 - Post Test Photograph of Stage 2 Cover

All other potential influences on rotor thrust have been considered in this analysis as well. The impulse of the gas and liquid streams on the impellers is included, and only serves to unload the rotor. The magnitude of the impulse force is very small in comparison to the pressure-area forces described above. The potential for hydrodynamic forces developing between the rotating and stationary components is also considered, but there is not tapered geometry in the direction of travel with which to develop a significant pressure wedge. The last influence, which is included by way of the measured pressures, is this increase in head due to the wet gas conditions. Figure 19 shows how the compressor pressure ratio varies when it operates with a certain amount of liquid phase. Data comes from tests done at a rotational speed of 8000 rpm at increasing LVF. The dry gas tests are plotted together with wet tests for a rapid comparison as a baseline. Particularly, one can note that when increasing the LVF, the operating point moves towards the high-left part of the PR-Q plane. This behavior happens for all the three CV positions in a quite linear shape. The slope of this movement is increasing at lower flow coefficient.

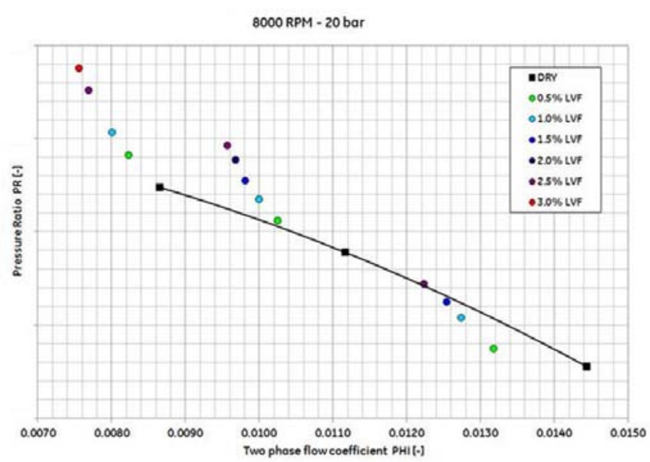

Figure 19 - Pressure Ratio vs. Gas Flow Rate

The pressure ratio raise can be explained with the combination of a series of effects. First of all the two-phase flow has a greater density than the dry air flow. There is also a cooling effect of liquid phase due to its incompressibility. Additionally, it is known in literature that the speed of sound for an air-water two phase flow decreases significantly even with small values of LVF. A simple model describing the two phase speed of sound can be found in [3], the so called Wood' law.

$$a = \frac{a_g}{1 - LVF}\sqrt{\frac{1 - LVF}{1 + LVF(r - 1)}}$$

Figure 20 shows the speed of sound calculated with the Wood law for the air-water two phase flow at 20 bar and 40°C, which are typical thermodynamic conditions of the flow at the compressor suction.

From the compressor point of view, the decreasing of the speed of sound, fixed the rotational speed, means that the peripheral Mach number is growing with LVF. Using the Wood law together with some proprietary nondimensional equations it is possible to predict the increasing of pressure ratio in a good way. Thus, the speed of sound variation plays a significant role in the compressor wet gas operations.

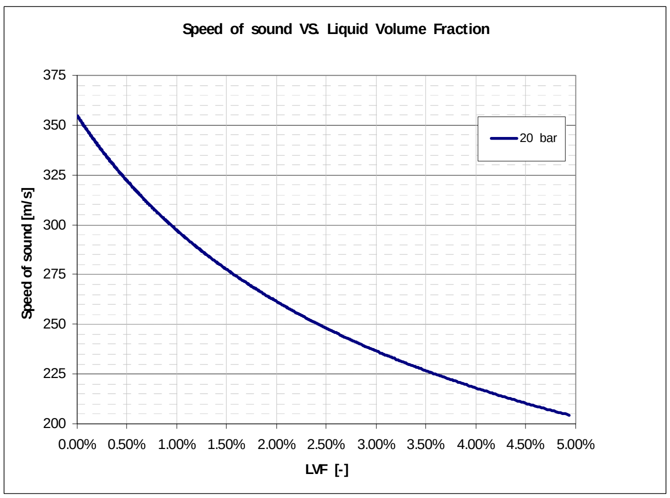

Figure 20 - Two-phase speed of sound vs. LVF. Values are calculated with the Wood' law for the compressor suction conditions

On the other hand, there is another effect that makes the pressure ratio decrease: the losses variation, whose effect is more important at highest flow rates, where the velocity is maximum. Therefore, this last point should explain how the slope of the dry-wet movement in Figure 19 is decreasing at highest gas flow rates.

CONCLUSIONS

This paper documents the successful test of a two-stage compressor in wet gas conditions with a suction pressure of 20 bar and a mixture of air and water volumetric flows up to 5% liquid volume fraction. One of the objectives of the test program is to characterize the machine mechanical performance under such conditions. As a result of the test program, the following conclusions are made:

1. Wet gas operation had negligible influence on the radial vibration. However, it is important to note that the test compressor contains worn eye and shaft seals, so the rotordynamic influence of two-phase flow in the seals may be much more severe for a machine with new, more restrictive seals.
2. Wet gas operation had a significant impact on the axial vibration. Two-phase flow in the balance piston seal served to reduce the thrust load at low LVF, which is an important consideration in the design of a compressor for wet gas operation. At high LVF, a new pressure cavity becomes an important contributor to the net thrust, but this may or may not be typical due to the worn condition of the eye seals.
3. Wet gas operation appears to also be related to the persistent 15 Hz axial vibration, although no further explanation is available for this measured vibration.
4. Pressure ratio increase during wet gas compression also contributes to the increased thrust load, and is directly related to the increasing Mach number at the stage discharge.

These findings will support the development of wet gas compressor thrust management design guidelines to prevent both the unloaded condition as well as the potential thrust bearing overload condition.

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